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METHOD AND DEVICE FOR DETECTING LEAKS IN RECIPROCATING

MACHINERY

This invention regards a method of detecting a leak in a
piece of reciprocating machinery. The method may also be used
5   for localising a leak. In this context, reciprocating
machinery means all types of pumps and hydraulic machineries
that are equipped with a rotating crankshaft or cam, where
the crankshaft or cam drives or is driven by at least two
pistons in a controlled reciprocating motion, and where each
10   piston cylinder is provided with at least two check valves
designed to rectify the direction of flow through the
machine. The invention also comprises a device for
implementing the method.

When operating reciprocating machinery, safety considerations
15   and economics make it very important to detect leaks in e.g.
piston seals and valves at an early stage. Leaks of this type
are acceleratory, and when they become large enough for the
operator of the reciprocating machinery to detect them
through large abnormal pressure variations, the reciprocating

machinery must often be shut down and overhauled immediately
and without the option of postponing the maintenance work to
a later and operationally more suitable time.

The object of the invention is to present a method of
5 detecting an incipient leak before it reaches an extent that
interferes with the operation, whereby repairs may be
scheduled for a later point of time.

The object is achieved in accordance with the invention by
the characteristics given in the description below and in the
10 appended claims.

The method comprises an analysis of the flow rate into and
out of the reciprocating machinery in order to detect a new
flow rate component, where the flow rate component has a
frequency that differs from the fundamental frequency of the
15 reciprocating machinery. Fundamental frequency is here taken
to mean the rotational frequency of the reciprocating
machinery multiplied by the number of pistons in the machine.
The flow rates may be estimates based on pressure
measurements.

20 Ideally, the flow rates into and out of the reciprocating
machinery should be as steady as possible, but in practice
the flow rates, and consequently the pressure, will fluctuate
in correlation with the speed of rotation of the machinery.
Such fluctuations are principally caused by geometric factors
25 that cause the sum of the piston speeds in each phase not to
be constant, the compressibility of the fluid which makes it
necessary to compress and then decompress the fluid prior to
equalising the pressure and opening the respective valves,

valve inertia that cause further delays in the opening and
closing of valves, and flow-dependent pressure drop through
valves and feed passages. If all pistons and valves are
identical and operate normally, the fluctuations will as a
result of symmetry have a fundamental frequency equal to the
rotational frequency of the machinery multiplied by the
number of pistons in the machine. However, if an abnormal
leak were to occur in e.g. one of the pistons or one of the
valves, the symmetry would be broken and the fluctuations in
the flow would have new frequency components, with the lowest
frequency equal to the rotational frequency of the machine.

The angular position θ of the rotating crankshaft or cam of
the reciprocating machinery is measured directly or
indirectly and normalised to values of between 0 and 2π,
optionally between $-\pi$ and π radians, where 0 represents the
start of the power stroke of piston no. 1. The reciprocating
machinery comprises two or more pistons uniformly
distributed, so that piston no. i of a total of n pistons has
a phase lag (angle) of $(i-1)2\pi/n$ relative to the first
piston.

The pressure into or out of the machinery is measured,
whereupon the measured pressure signals are transformed into
functions representing normalised flow rates into and out of
the machinery. This means that if, for instance, q_{in} and q_{av}
represent instantaneous and average flow rates into the
reciprocating machinery, the normalised inflow function is
defined so that

$$f/(p_{in}) \approx q_{in}/q_{av}.$$

It is well known to those skilled in the art that a periodic
signal can be split up into components of different
frequencies, e.g. by use of Fourier analysis. The ith harmonic
component of the reciprocating machinery frequency may be
represented by two coefficients defined by the following
integrals:

$$s_i = \frac{1}{\pi} \int_0^{2\pi} f \sin(i\theta)d\theta \qquad\qquad (1)$$

$$c_i = \frac{1}{\pi} \int_0^{2\pi} f \cos(i\theta)d\theta \qquad\qquad (2)$$

These integrals, which in practice must be implemented as
summations in a computer or in a programmable logic
controller (PLC), are updated for each new revolution of the
reciprocating machinery. If the function f is purely periodic
with the pump angle, the integrals will be constant. However,
f can often contain significant non-periodic components
caused by e.g. a change in the pump speed or a variation in
the external flow resistance. The non-periodic components may
be regarded as stochastic noise, and may consequently be
suppressed relative to the periodic components by use of
known smoothing or averaging techniques. As an example,
recursive smoothing can be applied according to the recipe

$$a_i(k) = (1-x)a_i(k-1)+xs_i(k) \qquad\qquad (3)$$

$$b_i(k) = (1-x)b_i(k-1)+xc_i(k) \qquad\qquad (4)$$

where the argument refers to the estimate following revolution number k of the machinery, and x is a positive weighting function that is normally much smaller than 1. This smoothing filter, which is a low-pass filter of the first order, allows all the previous estimates of sine and cosine coefficients to form part of the estimated resultant coefficients, but with the weighting decreasing exponentially backwards in time. If the non-periodic variations are relatively large, it may be necessary to use a second-order smoothing and/or select an even smaller weighting function x.

The amplitude of the first harmonic frequency component:

$$f_1 = \sqrt{a_1^2 + b_1^2} \qquad\qquad (5)$$

represents a quantitative measure of the leakage in the machinery. In the event of a small leak, this component will start to increase from its low normal noise level long before the leak becomes large enough for the associated pressure fluctuation to show up directly on a manometer or in a graph of the pressure signals.

The method is also well suited for locating the main source of the leak. This is possible because the Fourier analysis contains phase information through the two frequency components a_1 and b_1. The connection between the frequency components may through use of a well known trigonometric identity be expressed as

$$a_1 \sin(\theta) + b_1 \cos(\theta) = f_1 \sin(\theta - \varphi) \qquad\qquad (6)$$

where

$$\tan(\varphi_1) = a_1/b_1 \qquad\qquad (7)$$

The sign of the coefficient determines in which quadrant the angle φ_1 lies.

5   Each type of leak has its own signature, as explained in some of the following examples. The functions f and g represent functions that are approximately proportional to the incoming and outgoing volumetric flow rates respectively. This means that the associated phase angles φ_1 and γ_1 are direct

10   expressions of the first harmonic component of the maxima of the fluid flows.

The following describes several non-limiting examples of use of the method illustrated in the accompanying drawings, in which:

15   Figure 1 schematically shows a triplex pump equipped with the required measuring devices and analysers; and

Figure 2 shows a curve illustrating the delivered volumetric flow as a function of the rotational angle of the pump, showing the central point of a piston leak.

20   In the drawings, reference number 1 denotes a so-called triplex pump equipped with three individually acting pistons 2, 2' and 2" respectively, of which only piston 2 is shown, extending through their respective cylinders 4, 4' and 4". The cylinders 4, 4' and 4" communicate with an inlet manifold

6 through their respective inlet valves 8, 8' and 8", and an
outlet manifold 10 through their respective outlet valves 12,
12' and 12", respectively. An inlet pressure sensor 14 is
connected to the inlet manifold 6, communicating with a
5   computer 16 via a line 18, and an outlet pressure sensor 20
is connected to the outlet manifold 10, communicating with
the computer 16 via a line 22. A rotational angle transmitter
24 is designed to measure the rotational angle of the
crankshaft 26 of the pump 1, and is communicatingly connected
10 to the computer 16 by means of a line 28. The sensors 14 and
20, the transmitter 24 and the computer 16 are of types that
are known *per se*, and the computer is programmed to carry out
the calculations concerned.

In the event of a leak in the seal of the first piston 2, the
15 outflow through the outlet valve 12 is reduced during the
pumping phase by a quantity equal to the leakage past the
piston 2. As the pump stroke extends over half a revolution
of the crankshaft 26 of the pump 1, the central point 32, see
figure 2, for this reduction in volume flow is approximately
20 $\pi/2$ radians (90°) after the start of the pump stroke. In
figure 2, the curve 34 indicates the reduction in the average
volume flow 36 which occurs as a result of the piston
leakage. In reality, the central point 32 will lag by an
additional, small angle. This is due to both compression and
25 decompression of the pumped fluid, and the fact that with a
crank drive, where the crankshaft (not shown) of the piston 2
has a finite length, the maximum piston speed is reached only
at approximately 100°. These effects may be calculated and
compensated for by adding a pressure and geometry dependent
30 phase lag δ. The corresponding angle γ_1 on the first harmonic
component of the outflow function g will therefore be shifted

by half a revolution relative to that point, i.e. to $\gamma_1 = -\pi/2+\delta$. Due to the fact that the pressure during the return or suction stroke of the piston 2 is far lower, the inflow rate, and consequently also the function f, will be relatively unaffected by a piston leak, whereby $f_1 \approx 0$.

The above example may easily be generalised to a rule which states that in the event of a leak in piston number i of a total of n pistons with a mutual displacement of phases of $2\pi/n$, then

$$f_1 \approx 0 \tag{8}$$

$$g_1 > 0 \tag{9}$$

$$\gamma_1 = -\pi/2+\delta+2(i-1)\pi/n \tag{10}$$

The inequality > 0 here indicates that g_1 must be interpreted as being significantly greater than the level of background noise.

If the leak is in the inlet valve 8 instead of in the piston 2, the effect on the out function will be the same as in the case of a pure piston leak. However the drop in the outflow will now be balanced by a similarly negative inflow. It can therefore be concluded that in the event of a leak in in-valve number i, (9) and (10) will still apply, but (8) must be replaced by

$$f_1 > 0 \tag{11}$$

$$\varphi_1 \;=\; \gamma_1 \;=\; -\pi/2 + \delta + 2(i-1)\pi/n \qquad\qquad (12)$$

A leak in outlet valve 12 will lead to a reduction in both
the inflow and the outflow during the return phase. Then both
(11) and (9) will apply, but the phase angle is displaced by
half a revolution, to

$$\varphi 1 \;=\; \gamma 1 \;=\; \pi/2 + \delta + 2(i-1)\pi/n \qquad\qquad (13)$$

In those cases where n is an odd number, the source of the
leakage will be determined uniquely. The reason for this is
that the phase angles defined by equation (12) are all
different from the phase angles defined by equation (13).
However, if the number of pistons is an even number, the
leakage determination will be two-valued, at least if the
leak is in one of the valves. The reason for this is that a
leak in an inlet valve number i will have the same signature
as a leak in the complementary out valve (number $i \pm n/2$) and
vice versa.